

Innovative combined heat and power system based on a double shaft intercooled externally fired gas cycle



P. Iora^a, P. Silva^{b,*}

^a Università di Brescia, Dipartimento di Ingegneria Meccanica e Industriale, Via Branze 38, 25123 Brescia, Italy

^b Politecnico di Milano, Dipartimento di Energia, Via Lambruschini 4, 20156 Milano, Italy

HIGHLIGHTS

- A new CHP solution: a double shaft intercooled gas cycle with external combustion.
- Turbocharger technology from automotive industry has been exploited.
- Thermodynamic calculations are based on real turbochargers maps.
- Technical feasibility of a 50 kW biomass system with TIT of 750 °C is discussed.
- Electric efficiency of 21% is achieved, higher than ORC plants of this size.

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ABSTRACT

This paper presents a new CHP solution based on a double shaft intercooled gas cycle with external combustion (EFGT cycle). This configuration exploits the turbocharger technology widely diffused in the automotive industry, taking advantage of the capital cost reduction due to the economy of scale typical of the automotive market. Thermodynamic calculations based on actual turbochargers maps available from manufacturers data are carried out in order to evaluate the performances of the system. It is shown that the system results competitive both with natural gas fuelled solutions such as internal combustion engines and microturbines achieving conversion efficiency of 25–30%, as well as with low grade fuels like biomass. For a 50 kW biomass system with a grate combustor boiler and turbine inlet temperature of 750 °C, an electric efficiency of 21% is obtained, which is higher than the available state of the art solutions based on ORC technology for this size. A preliminary design of the system is performed, including the sizing of the boiler, showing its technical feasibility, while complete economic evaluations will be considered in future works.

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1. Introduction

The liberalization of the electric market, public environmental policies and the expanding demand for power have led in the last decades to a partial transition to diffused energy generation and cogeneration systems. Within this context the nominal power of combined heat and power (CHP) plants typically ranges from a few 10 kW up to 5 MW, and energy conversion devices such as micro-gas turbine (MGT) and internal combustion engine (ICE) can be considered the mainstream technologies.

Although the term “micro-gas turbine” has not yet been standardized, as a common assumption it refers to small scale energy production devices (<500 kW), based on recuperative gas cycles. This technology provides a solution to the scaling-down of higher

power gas turbines which, for power sizes under 5–10 MW, exhibit expensive capital costs and lower efficiencies compared with their competitors internal combustion engines. Starting from the middle 1990s, micro-gas turbine technology has been welcomed with enthusiasm by the scientific community and by the industry. However only few models are available today on the market, ranging from 30 to 250 kW with efficiency close to 30% [1,2]. ICEs, ranging from 10 kW to 5 MW, are mature market products and show, for a fixed nominal power, efficiencies generally higher than those typically obtained with MGTs. On the debit side ICEs have higher emissions and higher maintenance costs than MGTs.

In the power range of few 10 kW the specific capital costs of both ICEs and MGTs are still high, with values ranging from 1500 €/kW (for the highest size in the considered power interval) up to 5000–8000 €/kW in case of ICEs with power less than 10 kW [3]. Similarly, at these relatively low sizes, high costs during operation are expected, due to the rather low conversion efficiencies.

* Corresponding author. Tel.: +39 0223993889; fax: +39 0223993913.

E-mail address: paolo.silva@polimi.it (P. Silva).

Nomenclature

CHP	combined heat and power
EFAH	externally fired air heater
EFGT	externally fired gas turbine
EFHAT	externally fired humid air turbine
HP	high pressure
ICE	internal combustion engine
LP	low pressure
MGT	micro gas turbine
ORC	Organic Rankine Cycle
TIT	turbine inlet temperature
β	pressure ratio

η	isentropic efficiency
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Subscripts

APT	auxiliary power turbine
in	inlet conditions
LPC	low pressure compressor
LPT	low pressure turbine
HPC	high pressure compressor
HPT	high pressure turbine

Their competitiveness in a deregulated energy distribution market is therefore hindered, even when they operate in cogenerative mode with complete thermal recovery. Another characteristic feature of ICEs and MGTs is that they can be classified as internal combustion technologies, requiring high quality fuels such as natural gas. Biomass and other low quality fuels (synthetic gas, heavy oil, coal and garbage) can typically be employed in external combustion engines [4–8]. In this case the reference technology in distributed generation is the Organic Rankine Cycle (ORC). Commercially available ORC plants are in the range of 200 kW to few MW, showing conversion efficiency from 18% to 25% [9,10]. The overall capital cost of an ORC plant is presently high and can be assessed between 2500 and 4000 €/kW depending on the size of the system, so that the actual cost-effective size of a ORC plant is typically higher than 1 MW. Such an investment can be normally profitable only if some kinds of subsidies such as green certificates or feed in tariffs are available. It has to be noticed that for lower power CHP biomass plants, typically 1–100 kW, the Stirling engine [11,12] as well as technologies based on rotary steam engine [13] are available, although they still exhibit high investment costs and reliability constraints.

This work presents an innovative solution for distributed cogeneration, based on a gas cycle with external combustion. Externally fired gas turbines (EFGTs) are not a novel idea. The externally firing of a gas turbine entails that the hot combustion gases are not in direct contact with the turbine, and the working fluid of the cycle is separated from gases. Thermal power from the combustion gases is transferred to the working fluid by means of a high-temperature heat exchanger and the combustion generally takes place at atmospheric pressure, unlike conventional directly fired gas turbines. For these reasons externally fired cycles can deal with a wide range of fuels, even dirty or solid fuels such as biomass. On the contrary, in a direct fired gas turbine the only possibility of using biomass would be the gasification, with the adoption of complex and expensive devices. The externally fired gas turbine cycle can adopt two different configurations, namely with open or closed cycle [7]. In the closed cycle, the working fluid outgoing from the turbine is returned back to the compressor after a cooling process. In this case it is possible to choose among different fluids, with the possibility of optimizing both thermodynamic performances and heat transfer properties. As a drawback, the closed configuration yields high investment costs because of the number of the involved heat exchangers, as well as operational problems due to fluid leakages. In the open cycle the working fluid is air that from the outlet of the turbine is supplied to the combustion chamber. Since air temperature at this stage is generally higher than ambient temperature, a regenerative cycle is consequently obtained without any additional heat exchangers. Costs and reliability issues in most cases have led to the adoption of an open cycle in real applications [14], especially with solid fuels and biomass, as the case presented in this study.

Research on externally fired gas turbine systems has considerably increased during last years, especially in configuration fed by biomass. Some of these studies [15–17] have dealt with methods to increase the efficiency both in simple cycle and in combined cycle configurations, investigating the use of metallic as well as ceramic materials for high temperature heat exchangers. Another configuration capable of enhancing the power output and efficiency of the EFGT, by increasing the mass flow through the turbine, is the so called externally fired humid air turbine (EFHAT), which consists of an integration between the externally fired cycle with the evaporative gas turbine cycle [14,18,19]. Other arrangements of EFGT cycles exploit the available thermal power of the exhaust gases for biomass pre-drying processes [20,21], and even the use of organic waste fuels, like sewage sludge and poultry wastes, has been investigated [22]. Small power EFGT units for rural areas have also been studied [23].

This paper presents an innovative CHP solution for distributed generation, based on a double shaft intercooled and reheated gas cycle with external combustion. This configuration exploits the turbocharger technology widely diffused in the automotive industry, taking advantage of the capital cost reduction due to the economy of scale of the automotive market and also of the significant improvements in terms of efficiency, reliability and maximum temperature tolerable by the materials occurred in the past decades. Turbochargers' adoption has already been investigated both in internally fired cycles for automotive application [24] and in externally fired cycles fed by biomass [6,7]. This paper for the first time evaluates the performance of the cycle considering data derived from actual turbochargers maps available from automotive catalogs. Moreover, as a difference from similar works [4,7,23], a preliminary sizing of the high temperature heat exchanger is performed. The potential advantages of this configuration with respect to the existing technologies are discussed on the basis of numerical simulations. Finally the technical feasibility of the system is examined with reference to a 50 kW biomass fuelled plant, including the sizing of the boiler, while economic evaluations will be treated in future works.

2. Plant lay-out description

The system layout is shown in Fig. 1. Ambient air is compressed to the maximum plant pressure through a two stages intercooled compression (streams 1–4). The air flow is then heated to the maximum cycle temperature (TIT) in the high pressure section of the externally fired air heater (HP-EFAH) and then expanded in the HP turbine (streams 5 and 6). The air flow is then reheated to TIT in the LP-EFAH section and expanded in the LP turbine (streams 7 and 8). The compressors and turbines are coupled by means of HP and LP shafts, in the same way as turbochargers for automotive applications. In case both the HP and LP turbochargers are operated

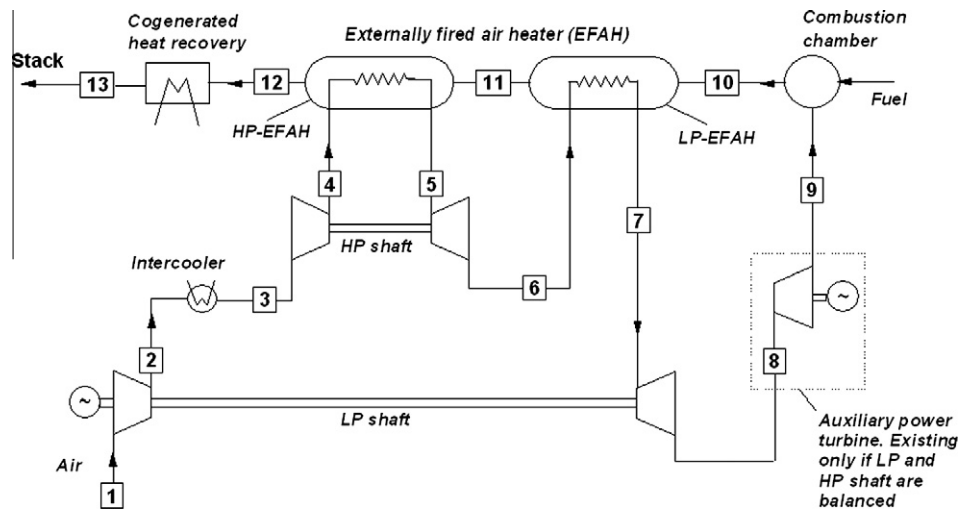


Fig. 1. System layout.

in balanced mode an auxiliary expander is included to complete the expansion process to conditions close to ambient pressure (streams 8 and 9), producing electricity. The exit flow is at atmospheric pressure and has still a relatively high temperature. Therefore it is introduced in the combustion chamber, providing also the necessary oxygen for the combustion of the fuel. The high temperature combustion products (stream 10) are employed to heat the air in the EFAH (streams 10–12) and the remaining heat available in the exhaust stream is recovered for cogeneration purposes (stream 12 and 13). Further heat can be recovered from the intercooler, depending on the temperature level required by the user.

Notably this layout can be classified as an external combustion system operating with air as the working fluid. The main features of this configuration are the following:

- The HP and LP turbochargers are available from the automotive sector in a wide variety of size and models, resulting in a broad spectrum of possible power scale of the system.
- The high pressure turbine and compressor are balanced while the low pressure can either produce power or operate in balanced mode if an additional expander is present. As it will be better explained in the next sections, the latter option can be beneficial as it allows both turbochargers to operate close to their nominal point.
- The EFAH is directly fed by the low pressure turbine outlet stream thus realizing a regenerative cycle avoiding the presence of a regenerator.
- In principle the external combustion allows to employ any kind of fuel, including biomass or other solid or low grade fuels.

3. Thermodynamic analysis

A thermodynamic analysis of the system shown in Fig. 1 is carried out with the aim of evaluating the plant performances in a fairly wide range of operating conditions. For simplicity at this stage the considered plant configuration is the one without the auxiliary expander, where the unbalanced LP shaft produces the plant power. Calculation assumptions are reported in Table 1.

The plant efficiency and the power output are determined as a function of compressor and turbine isentropic efficiency with reference to TIT values of 650, 750, 850 and 950 °C. The overall pressure ratio (β_{tot}) has been also optimized, and in all the considered

Table 1

Calculation assumptions for the system shown in Fig. 1.

$\Delta p/p$ Intercooler	2%
$\Delta p/p$ EFAH cold side	3%
$\Delta p/p$ EFAH hot side (overall)	5% [4]
$\Delta p/p$ cogenerated heat recovery heat exchanger	2%
Inlet and outlet pressure loss (streams 1 and 13)	1000 Pa [4]
ΔT_{min} EFAH	50 °C
Intercooler exit temperature (T_3)	35 °C
Ambient Temperature (T_1)	15 °C
EFAH combustion efficiency (included thermal losses)	93% [4]
HP shaft organic efficiency	98% [4]
LP shaft organic/electric efficiency (with inverter)	98/92% [4]

cases is between 3 and 5.5.¹ It can be noted that these values are compatible with specifications in terms of temperature and pressure for turbochargers typically adopted in the automotive sector. Results of the analysis are plotted in Fig. 2, suggesting the following comments:

- In case of TIT = 850 °C and TIT = 950 °C it is possible to obtain system efficiencies of about 25% and 28% respectively, adopting compressor and turbine isentropic efficiencies in the range of 75–77%. These figures can normally be found in typical turbochargers of the automotive industry. For this fairly high values of TIT, the system should be compared to the natural gas fuelled MGTs and ICEs in a power range of 30–250 kW. Although a potential reduction of costs can be envisaged, no substantial improvements in terms of efficiency can be demonstrated with respect to the state of the art of these technologies. It has to be noticed that with respect to ICEs, lower pollutants emissions should be reached, thanks to the external combustion process and the possibility of adopting a premixed low-NO_x natural gas combustor.
- The case of TIT = 750 °C shows a system efficiency of about 20% in correspondence of a compressor and turbine isentropic efficiency of 75–77%, while a marked efficiency reduction is observed in case of TIT = 650 °C. Considering that a TIT value of 750 °C can typically be obtained using low grade fuels such as biomass, the comparison in this case should be made with

¹ The same pressure ratio is assumed for LP and HP compressor i.e. $\beta_{HP} = \beta_{LP} = \sqrt{\beta_{TOT}}$. It can be noted that this is not necessarily the optimum condition, however an optimization of the values of β_{LP} and β_{HP} would yield only a marginal improvement on the resulting cycle efficiency.

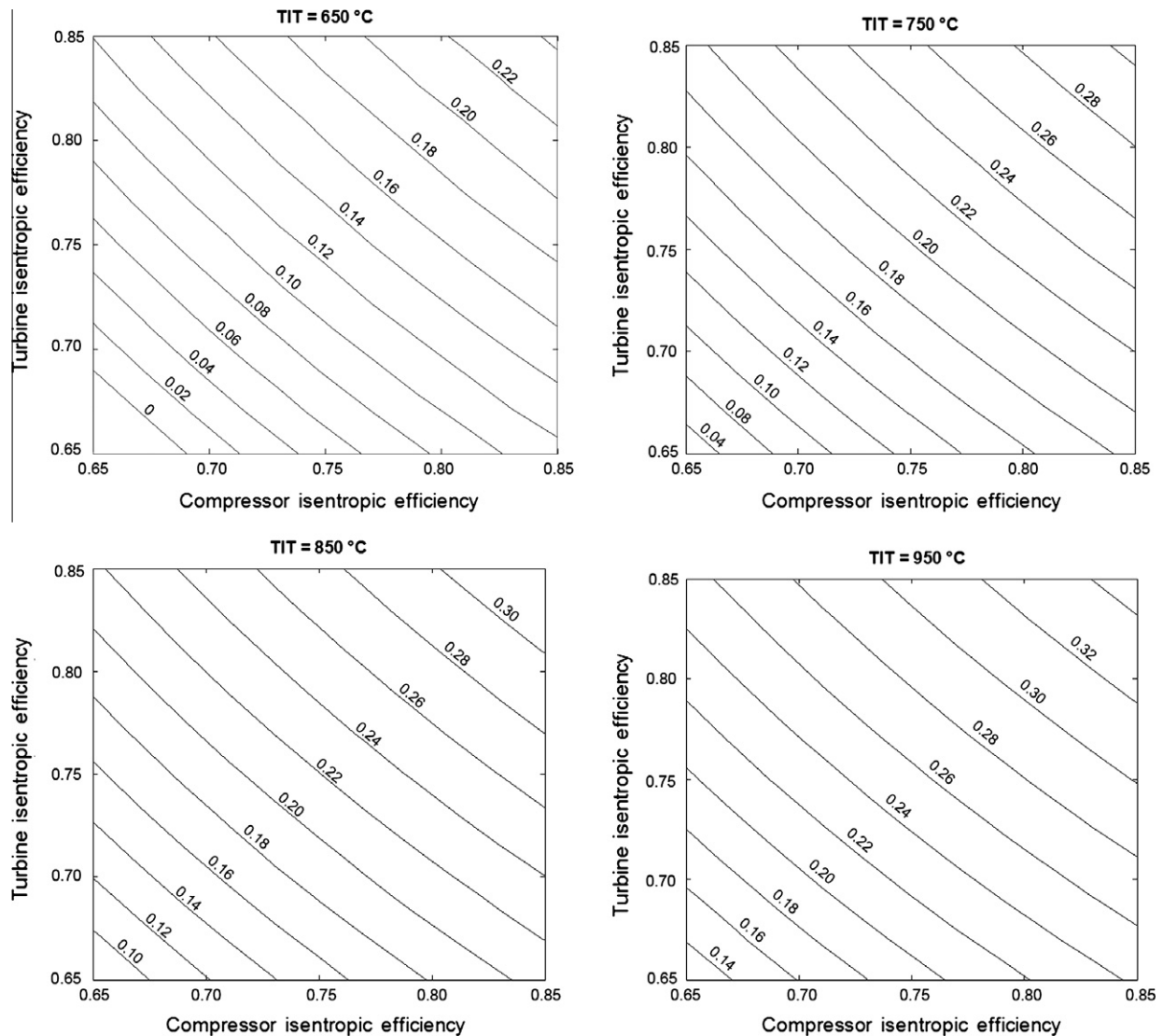


Fig. 2. Cycle efficiency as a function of compressor and turbine isentropic efficiency for four values of TIT (650 °C, 750 °C, 850 °C, and 950 °C).

the technology of external combustion ORC, where plant efficiency of about 18–19% can typically be achieved in cogenerative applications in the power range of few 100 kW, operating with TIT in the range of 250–300 °C compatible with the thermal stability of the typical working fluids employed such as hydrocarbons and siloxanes [9,10]. Potential cost reductions obtained with comparable or higher values of efficiency suggest deeper investigation of this application. Thus the next sections will particularly focus on this specific case.

4. Components selection

A preliminary design of the system shown in Fig. 1 requires an appropriate selection of the technology of the two turbochargers and the EFAH which are certainly the components of major concern in the power plant. Regarding the choice of turbochargers, although commercial products available for applications in automotive, gensets or naval sectors can cover a fairly wide range of power size, the target of this analysis is to consider power generation in the range of 50–100 kW where the present system can be particularly competitive in terms of performances and costs with the traditional technologies of ICE and MTG. Furthermore, this power size can hardly be developed with an ORC plant owing to

the intrinsic technological challenge of designing a turbine of comparatively low power, and the proportional increase of specific cost observed for plants with power less than 200 kW based on this technology [9,10].

The characteristic maps of the selected commercial turbochargers have been implemented in Aspen Plus in order to achieve a consistent simulation of the plant. They are shown in Figs. 3 and 4 [25].

It is important to remark that calculations based on the turbochargers maps shown in Figs. 3 and 4 yield poor results in terms of system power output and efficiency, as long as the layout configuration with power produced at the LP shaft is considered. This is because the turbochargers for automotive application considered in this study are designed to operate in balanced mode and show a significant reduction in performances when operated aside the design point. In addition to fluid-dynamic concerns, mechanical design criteria should be taken into account: together with TIT limitations, the most critical aspects to be considered are the axial thrust on the shaft, due to pressure differences between compressor and turbine, and bearing lubrication issues. These issues have to be carefully considered in the design process of the system as well as in the adaptation of existing turbochargers.

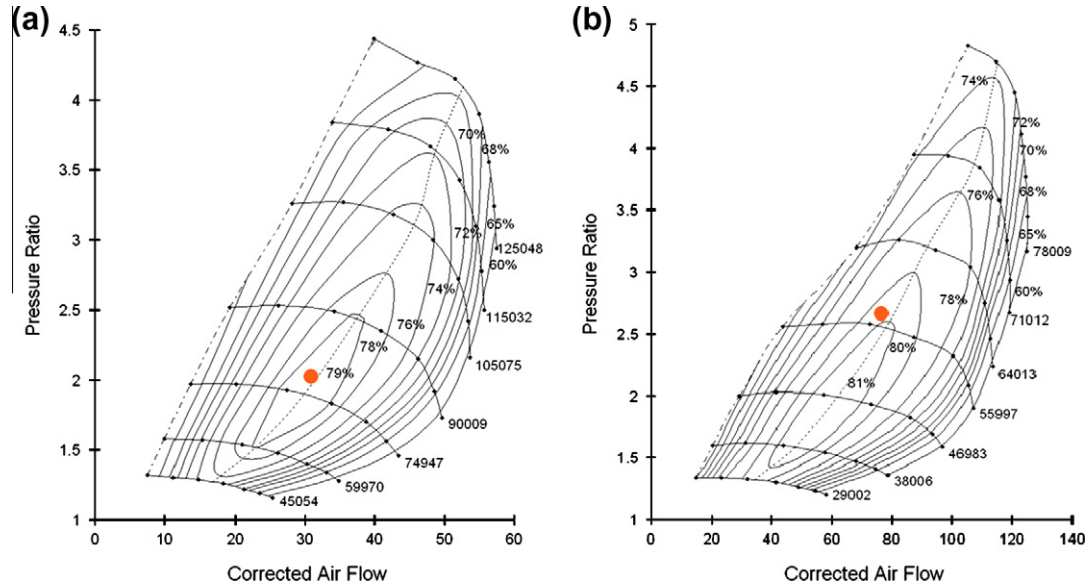


Fig. 3. High pressure compressor map (a) and low pressure compressor map (b) [23]. The actual air mass flow $\dot{m}$ and rotational speed N are related to the values of the corrected air flow $\dot{m}_c$ and corrected speed N_c reported in the diagrams by the following expressions: $\dot{m}_c = \dot{m} \frac{\sqrt{T_{in}/545}}{P_{in}/28.4}$ and $N_c = \frac{N}{\sqrt{T_{in}/545}}$ where T_{in} is in Reaumur, P_{in} in inch of Hg, $\dot{m}$ in lib/min and N in rpm. The red dots denote the operating condition in case of TIT = 750 °C (see Section 5).

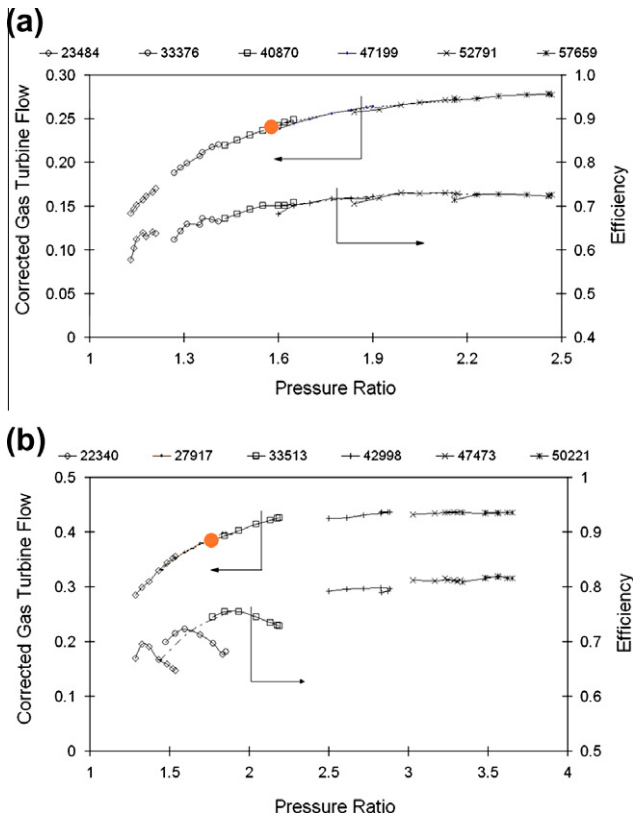


Fig. 4. High pressure turbine map (a) and low pressure turbine map (b) [23]. The actual air mass flow $\dot{m}$ and rotational speed N are related to the values of the corrected air flow $\dot{m}_c$ and corrected speed N_c reported in the diagrams by the following expressions $\dot{m}_c = \dot{m} \frac{\sqrt{T_{in}/519}}{P_{in}/29.92}$ $N_c = \frac{N}{\sqrt{T_{in}/519}}$ for the high pressure map (a) and $\dot{m}_c = \dot{m} \frac{\sqrt{T_{in}/288}}{P_{in}/101325}$ $N_c = \frac{N}{\sqrt{T_{in}/288}}$ for the low pressure map (b). In both cases T_{in} is in Reaumur, P_{in} in inch of Hg, $\dot{m}$ in lib/min and N in rpm. The red dots denote the operating condition in case of TIT = 750 °C (see Section 5).

It is worth observing that modified turbochargers are presently commonly used in refrigeration air cycles, especially in aerospace applications. Cabin air conditioning of civil and military aircrafts is in fact generally obtained by means of a so called “bootstrap cycle”, based on turbochargers that exploit cooling through the expansion of compressed air taken from the engines. A discussion of this type of systems can be found in [26]. Turbochargers derived from industrial mass-production have been already used in different applications: some papers discussing the potential use of 2modified turbochargers for air conditioning in buildings [27], refrigeration for road transport vehicles [28], and cooling for drying process [29] are available. In most cases original plain oil-fed journal bearings are maintained, showing thrust capacity of the plain bearing arrangement even when working at pressures different from original values [28]. Other papers can be found discussing the unstable operating conditions (stall or surge) of automotive turbochargers [30], or moisture effects during air expansion [31]. The use of turbochargers in biomass EFGT cycles has been already investigated and tested in [6,7], also discussing start-up issues.

The application proposed in this paper shows similar features as far as the use of modified turbochargers is concerned. Considering both LP or HP shaft (the most critical component), axial thrusts are still balanced, as in the original automotive case. The adoption of balanced turbochargers prevents the occurrences of axial thrusts and should avoid the re-design of the shaft bearings, while the bearing lubrication should be guaranteed through an increase of oil pressure and without any modification to the bearings [7].

For these reasons we consider the configuration with power produced by the auxiliary power turbine, keeping both the turbochargers operating in balanced mode (Fig. 1), assuming that the auxiliary power turbine completes the expansion of the LP turbine to ambient pressure maintaining the same isentropic efficiency. It is worth to observe that the design of the auxiliary power turbine is a minor issue in the present application, given the relatively limited pressure ratio and fairly low inlet temperature. The idea to place an additional expander at the outlet of a turbocharger has already been explored in automotive engines. The technique is known in literature as turbocompounding and it is either used to

produce additional mechanical power or electricity to recharge batteries in a hybrid vehicle [32].

Regarding the design of the EFAH, it can be observed that several examples of high-temperature heat exchangers, used in externally fired gas turbine systems, are based on the shell and tube configuration [4–8]. In this case two sections of a shell and tubes heat exchanger arranged in series are considered. Details on the preliminary design of the EFAH will be provided in Section 6.

5. Simulations and results

Simulations are carried out with Aspen Plus adopting the calculation assumptions reported in Table 1. The LP compressor inlet mass flow was determined in order to maintain the turbochargers operation as close as possible to their nominal conditions, according to the maps shown in Figs. 3 and 4 for the selected machines. Results are shown in Table 2 for three different values of TIT. Thermodynamic data for each stream of the plant are listed in Table 3 in case of TIT = 750 °C. The analysis of the results suggests the following considerations:

- In all cases the turbochargers operate close to their design point in terms of rotational speed and pressure ratio with efficiency in the range of 70–80%.
- The overall pressure ratio is not equally split in LP and HP shafts as in the thermodynamic analysis, due to the adoption of real maps for compressors and turbines.
- The electric power of the system is in the range of 50–80 kW, depending on the TIT value. In particular for TIT = 750 °C, where the comparison should be made with the ORC technology, the plant produces 49.2 kW with an electrical efficiency of 20.7%. This result confirms the considerations made in the thermodynamic analysis, evidencing the superior performances in terms of efficiency and possibility of downsizing, with respect to the competing ORC technology. For higher values of TIT, power and efficiency are similar to figures obtained with ICEs and MGTs fuelled by natural gas.
- For lower TIT values the ratio between electricity and cogenerated thermal power is similar to ORC applications with biomass [9,10].

Table 2
Simulation results in case of three TIT values.

	TIT = 750 °C	TIT = 850 °C	TIT = 950 °C
Electric power (kW)	49.2	62.7	79.5
Thermal power (kW)	116.7	114.9	122.7
Electrical efficiency (%)	20.7	23.5	27.0
Thermal efficiency (%)	49.1	43.0	41.7
Temperature of the recovered heat (°C)	90	90	90
Overall efficiency (%)	69.8	66.5	68.7
LP compressor inlet mass flow (kg/s)	0.620	0.615	0.610
Overall pressure ratio	5.13	5.37	5.60
LP compressor pressure ratio	2.60	2.62	2.61
HP compressor pressure ratio	2.01	2.10	2.19
LP turbine expansion ratio	1.75	1.68	1.57
HP turbine pressure ratio	1.57	1.55	1.53
Auxiliary expander pressure ratio	1.61	1.77	1.96
LP shaft speed (rps)	917	919	918
HP shaft speed (rps)	1313	1344	1380
^a η_{LPC} (%)	80.5	80.5	80.5
^a η_{HPC} (%)	80	79.5	79
^a η_{LPT} (%)	74	73	72
^a η_{HPT} (%)	70	70	70
^b η_{APT} (%)	74	73	72

^a Resulting from turbocharger's operating points obtained from maps shown in Figs. 3 and 4.

^b Assumed equal to η_{LPT} .

- In all cases the overall efficiencies, sum of electric and thermal ones, are close to 70%.

It can be noted that similar results in terms of efficiency could be achieved by choosing different turbocharger models, thus allowing to select the plant size in a fairly wide range of power, according to the energy demand of the user, the availability of fuel or any other relevant aspects related to the system economy.

6. Design of the EFAH

This section deals with the preliminary design of the EFAH, which undoubtedly represents the most critical and expensive component of the system. It can be observed that since the maximum temperature of the cycle is limited to 750 °C, stainless steel alloys can be employed instead of the more expensive austenitic nickel–chromium based super-alloys, as reported in a recent analysis carried out in [5]. Also, Al-attab and Zainal [7] report on a high-temperature heat exchanger used in a biomass-fed externally fired gas turbine system fabricated using stainless steel (SS-304), that was proved suitable to achieve a turbine inlet temperature of 694 °C. Due to the slightly higher temperature ranges of the present case, we considered the stainless steel SS-304L, given its comparatively higher corrosion resistance at high temperature [34]. Further analysis will also take into account new generation ferritic stainless steel with high resistance to high temperature oxidation.

Examining the technical literature about recuperators in gas cycles, there are several examples of high temperature heat exchangers employing exhaust gases from different fuels [35–38], while innovative configurations and geometries improving heat transfer are considered in [39,40].

As far as the EFAH assembly is concerned, two separate shell and tubes heat exchangers, arranged in series are adopted, indicated as HP-EFAH and LP-EFAH in Fig. 1.

The preliminary design of the two components was carried out with Aspen Exchanger Design & Rating [41], assuming the stream temperature variations and the pressure losses as by Table 3. The resulting geometry and heat exchange parameters are reported in Table 4. It can be noted that since the design ΔT of the hot streams across the HP-EFAH and LP-EFAH are 595.5 and 70 °C respectively, the resulting heat duties of 419 and 54 kW determine a prevailing size and cost of the HP-EFAH. Regarding the allocation of the streams, the following general considerations can be made [42]: (i) the more seriously fouling fluid (the hot stream of the combustion products in the present case) normally flows inside the tubes, since the tube side is easier to clean; and (ii) high pressure fluid (the cold streams of air in the present case) normally flows through the tubes. In this case it can be argued that considering the series arrangement of the two shell and tubes elements, the cleaning issue will principally concern the first section (LP EFAH) that has significantly smaller surface (Table 4). Thus it was chosen to set the cold stream (air) flowing through the tubes and the hot stream (combustion products) flowing through the shell. This is also the approach followed by Al-attab and Zainal [7]. In future works a thorough analysis on the possible design options in terms of stream arrangements and employed materials will be considered, in order to identify the most cost-effective solution for the present application.

In addition to different configurations of the EFAH, a complete economic evaluation and detailed sizing of all the components will be considered in future works, together with off-design performance assessment and control issues. A preliminary market analysis suggests that, in order to be competitive the target price for the double shaft technology is around 4000 €/kW. This value seems to be reasonable, considering that the proposed cycle has a potentially

Table 3

Properties of significant streams in the flowsheet shown in Fig. 1. Case of biomass plant with TIT = 750 °C presented in Table 2..

Point	T (°C)	P (bar)	$\dot{m}$ (kg/s)	Flow composition (% mol)				
				CO ₂	H ₂ O	N ₂	O ₂	Ar
1	15.0	1.01	0.620	–	–	78.0	21.0	1.0
2	127.1	2.60	0.620	–	–	78.0	21.0	1.0
3	35.0	2.55	0.620	–	–	78.0	21.0	1.0
4	119.9	5.13	0.620	–	–	78.0	21.0	1.0
5	750.0	4.98	0.620	–	–	78.0	21.0	1.0
6	673.2	3.18	0.620	–	–	78.0	21.0	1.0
7	750.0	3.08	0.620	–	–	78.0	21.0	1.0
8	650.5	1.76	0.620	–	–	78.0	21.0	1.0
9	571.7	1.10	0.620	–	–	78.0	21.0	1.0
10	870.0	1.08	0.635	2.1	2.8	76.0	18.1	1.0
11	800.0	1.05	0.635	2.1	2.8	76.0	18.1	1.0
12	204.5	1.03	0.635	2.1	2.8	76.0	18.1	1.0
13	100.0	1.01	0.635	2.1	2.8	76.0	18.1	1.0
Biomass mass data [33]				Composition (mass basis): 49.3% C, 6.0% H, 44.2% O, 0.5% N				
				Lower heating value: 18.5 MJ/kg				

Table 4

Results of the preliminary design of the two EFAH elements.

	LP EFAH	HP EFAH
<i>Geometry</i>		
Shell outer diameter (mm)	460	790
Tubes length (mm)	1800	3600
Number of tubes	225	3080
Tubes outer diameter (mm)	19.05	10
Overall mass (kg)	750	4100
Overall heat exchange surface (based on external tubes surface) (m ²)	43.6	334
<i>Heat transfer</i>		
Heat duty (kW)	54.1	419
Overall heat transfer coefficient (based on external tubes surface) (W m ⁻² K ⁻¹)	43.6	19.6
Log mean temperature difference (K)	123	66.5
Log mean temperature difference correction factor [43]	0.44	0.96
Highest velocity (shell/tubes) (ms ⁻¹)	58.2/ 13	33.4/ 2.3

higher efficiency and can be applied to lower sizes than existing ORC applications. In fact, analyzing the commercially available ORC systems, it has been already pointed out in Section 1 that the smaller plant size in the range of 200 kW, has similar or lower efficiency and its specific cost is the range of 4000 €/kW. Also a preliminary sizing of EFAH for this configuration have led to a cost of about 90–120 k€, implying that about half of the cost is due to this component. Nonetheless this value is obtained for the design and manufacturing of a first of a kind EFAH unit, so that a potential reduction of costs should be expected in correspondence of economy of scale effects due to market penetration of the technology.

7. Conclusions

The work presents a new CHP solution based on a double shaft intercooled gas cycle with external combustion. This configuration exploits the turbocharger technology widely diffused in the automotive industry, taking advantage of the significant improvements in terms of efficiency and materials development occurred in the past decades. Turbochargers benefit also from capital cost reduction due to the economy of scale in the automotive market and have reached high level of reliability. The potential advantages of this solution with respect to the existing technologies have been discussed on the basis of numerical simulations carried out on actual turbochargers maps available from manufacturers data.

Application fields could be either natural gas cogenerative plant in the range of 10 kW or few 100 kW, achieving electrical efficiency up to 30% and potential emissions reduction with respect to the state of the art technology of ICEs, or biomass fuelled plants where the reference technology is represented by ORC. The technical feasibility of a 50 kW biomass system with a grate combustor boiler and turbine inlet temperature of 750 °C has been discussed in details, showing an electric efficiency of about 21%, which is higher than the available state of the art solutions based on ORC technology for this size. A preliminary design of the system has been performed, including the design and sizing of the boiler, showing its technical feasibility, while complete economic evaluations will be considered in future works.

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